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Robert H. Hart

THE DESIGN AND PRELIMINARY TESTING  
OF A QUICK OPENING VALVE

187

A THESIS

Presented to  
the Faculty of the Division of Graduate Studies  
Georgia Institute of Technology

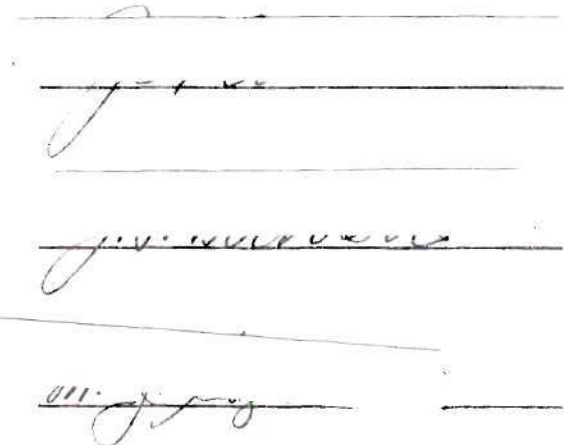
In Partial Fulfillment  
of the Requirements for the Degree  
Master of Science in Mechanical Engineering

by  
Robert Harris Hart

June 1953

THE DESIGN AND PRELIMINARY TESTING  
OF A QUICK OPENING VALVE

Approved:

Four handwritten signatures are written on four horizontal lines. The first signature is a simple horizontal stroke. The second signature is a cursive 'J' followed by a horizontal line. The third signature is a cursive 'J.V.' followed by a horizontal line. The fourth signature is a cursive 'J.V.' followed by a horizontal line.

Date Approved by Chairman: May 30, 1953

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## SYMBOLS

$\bar{A}_p$	Acceleration of piston (constant acceleration).....	ft/sec <sup>2</sup>
$A_c$	Clearance area between cushion wall and piston.....	in <sup>2</sup>
$A_p$	Cross-sectional area of piston.....	in <sup>2</sup>
$A_l$	Cross-sectional area of linkage and poppet stem.....	in <sup>2</sup>
$A_r$	Cross-sectional area of enlarged section of poppet stem.....	in <sup>2</sup>
$a_r$	Acceleration of poppet (constant acceleration).....	ft/sec <sup>2</sup>
$b$	Radial clearance between piston and cushion wall.....	in
$C_d$	Coefficient of discharge.....	
$E$	Modulus of elasticity.....	lbf/in <sup>2</sup>
$f$	Overall stress concentration factor	
$F_{pr}$	Average spring force, poppet-return spring.....	lb <sub>f</sub>
$F_{sr}$	Piston-return spring force.....	lb <sub>f</sub>
$F_{sri}$	Initial load, piston-return spring.....	lb <sub>f</sub>
$g$	Gravitational constant.....	32.2 ft/sec <sup>2</sup>
$K$	Dimensional constant.....	$2.31 \frac{\text{ft}(\text{H}_2\text{O})\text{in}^2}{\text{lb}_f}$
$K_{sr}$	Spring rate, piston-return spring.....	lb <sub>f</sub> /in
$L$	Length.....	in
$M_p$	Mass, piston.....	$\frac{\text{lb}_f \text{ sec}^2}{\text{ft}}$
$M_r$	Mass, poppet and attached link.....	$\frac{\text{lb}_f \text{ sec}^2}{\text{ft}}$

## SYMBOLS (continued)

Pc	Pressure, water cushion.....	lb <sub>f</sub> /in <sup>2</sup>
Pda	Pressure, driving air.....	lb <sub>f</sub> /in <sup>2</sup>
Q	Flow rate of water from cushion.....	in <sup>3</sup> /sec or ft <sup>3</sup> /sec
S	Stress.....	lb <sub>f</sub> /in <sup>2</sup>
T	Time.....	sec
Vc	Velocity, water escaping from cushion..	ft/sec
Vp	Velocity, piston.....	ft/sec
Vpo	Velocity, piston at instant of striking water cushion.....	ft/sec
X	Distance.....	in
Y	Penetration into cushion.....	in



THE DESIGN AND PRELIMINARY TESTING  
OF A QUICK OPENING VALVE

(34)

Robert Harris Hart

The problem considered in this thesis was the design of a valve which would meet the following requirements:

1. The release of measured quantities of compressed gas,
2. The opening time of the valve to be controllable over the range from  $1/4$  to 1 millisecond, and
3. The released gas not to be permitted to expand prior to the time of release.

The valve designed and built was a pilot-operated, piston-type valve. The pilot valve was triggered by a solenoid which lifted a poppet valve off its seat. The poppet opened a port and permitted the driving air pressure to reach the head end of the piston. The piston was accelerated by the driving air through a pretravel distance so that the piston velocity would be high during the uncovering of the discharge port. The driving air pressure is the control on the opening rate of the valve.

Upon completion of the opening of the discharge port the piston has a high velocity and a high kinetic energy.

The piston was arrested by a water cushion which dissipated this energy without injury to the parts of the valve.

The valve was designed using approximations which are usual in the design of mechanical devices. Subsequent to the design of the valve, a more detailed analytical consideration was made of the motion of the piston and of the action of the water cushion. The results of these considerations were in agreement with the design of the valve and substantiate the approximations made.

The results of the tests made with the valve indicate that the valve performed as predicted. The instrumentation used in the tests was not sufficiently responsive to permit exact determination of the motion versus time relationships of the various parts.

A curve showing the opening time of the discharge port versus the driving air pressure was plotted from the detailed consideration made of the piston motion. This curve is to be used in predicting the opening time of the valve.

The test results indicate that the opening time of the valve is in the range required and is controllable over this range. The valve, as designed, meets the requirements as given above and is considered a satisfactory solution to the design problem.

# THE DESIGN AND PRELIMINARY TESTING OF A QUICK OPENING VALVE

## CHAPTER I

### INTRODUCTION

So-called quick-opening valves have been built for many years. The term "quick-opening valve" has come to be applied to any valve which has an opening time that seems short to the operator. This concept is generally satisfactory to the trade and is applied to many types of hand- and motor-operated valves. In certain cases, such as fast-response pneumatic systems, the actual time interval for operation of the valve becomes more critical, and more careful consideration of operating time becomes necessary. The valve\* discussed in this thesis was built to provide an unusually fast release of compressed gas. The specific need for this valve is currently classed as security information and may not be stated in this presentation. The downstream situation will not be discussed other than to list the limitations on the opening time of the valve and the volume of gas to be released.

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\*The valve which comprises part of the subject matter of this thesis may be covered in an application for patent.

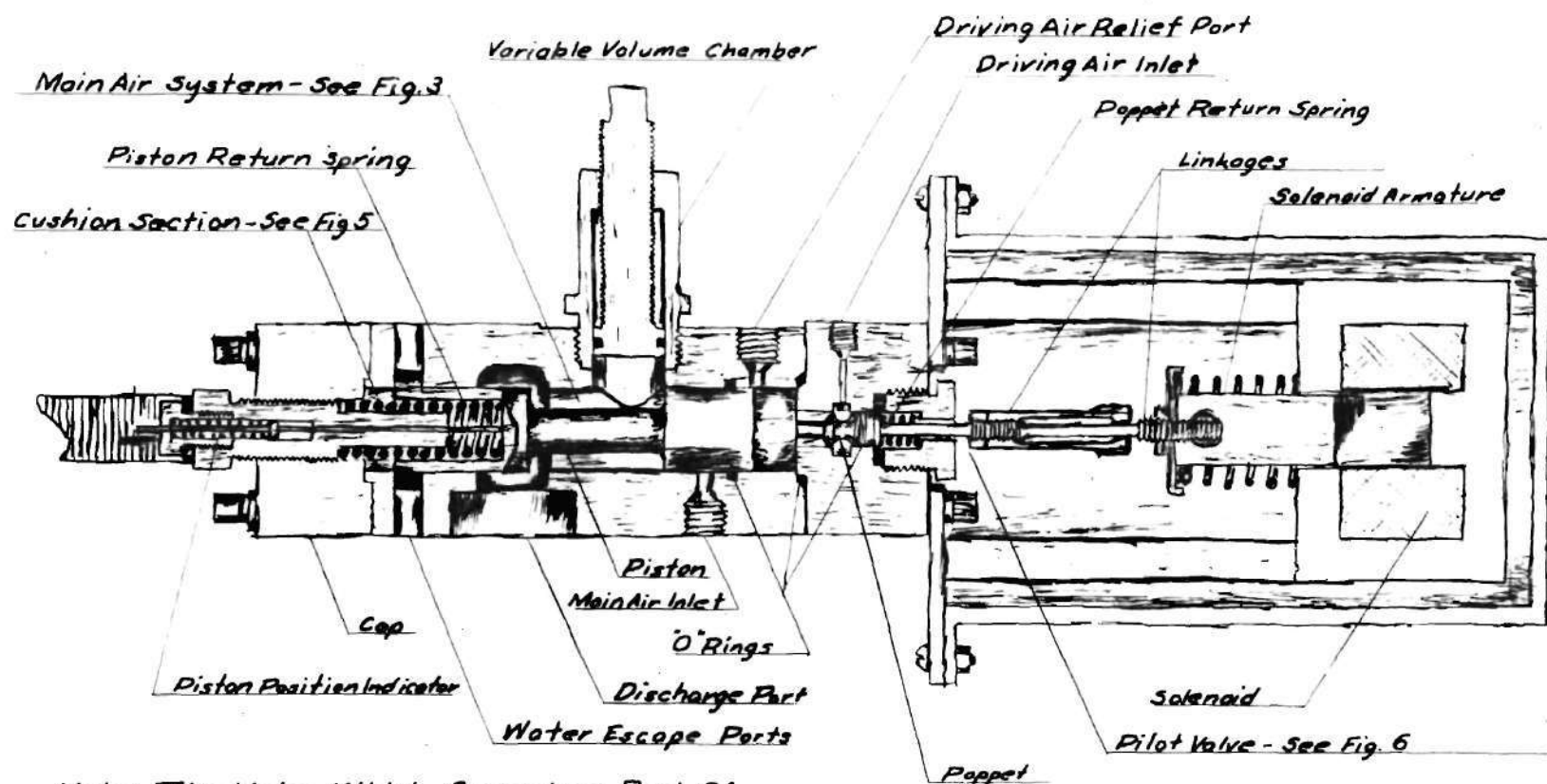
## CHAPTER II

### DESIGN REQUIREMENTS

These are the specific requirements to be met by the valve:

1. The interval of time required for opening must be controllable in the range from  $1/4$  millisecond to 1 millisecond.
2. The valve must release a measured volume of gas without permitting expansion of the gas prior to its release. The measured volume must be variable from 1.8 to 3.0 cubic inches.

The valve as designed and constructed (Figs. 1 and 2) is a piston-type valve. The piston is driven through its stroke by the driving air, which is admitted to the head end of the valve cylinder by the pilot valve. As the piston moves through its stroke, it closes the charging port and isolates the gas charge in the variable-volume chamber. The piston continues its movement, uncovering the discharge port and releasing the isolated gas charge. The moving piston is stopped by a water cushion and returned to its original position by the piston-return spring. The pilot valve is triggered by the solenoid and admits driving air to the head end of the cylinder.



Note: The Valve Which Comprises Part Of  
 The Subject Matter Of This Thesis May  
 Be Covered In An Application For Patent

Figure 1. Cross-section of Air Valve



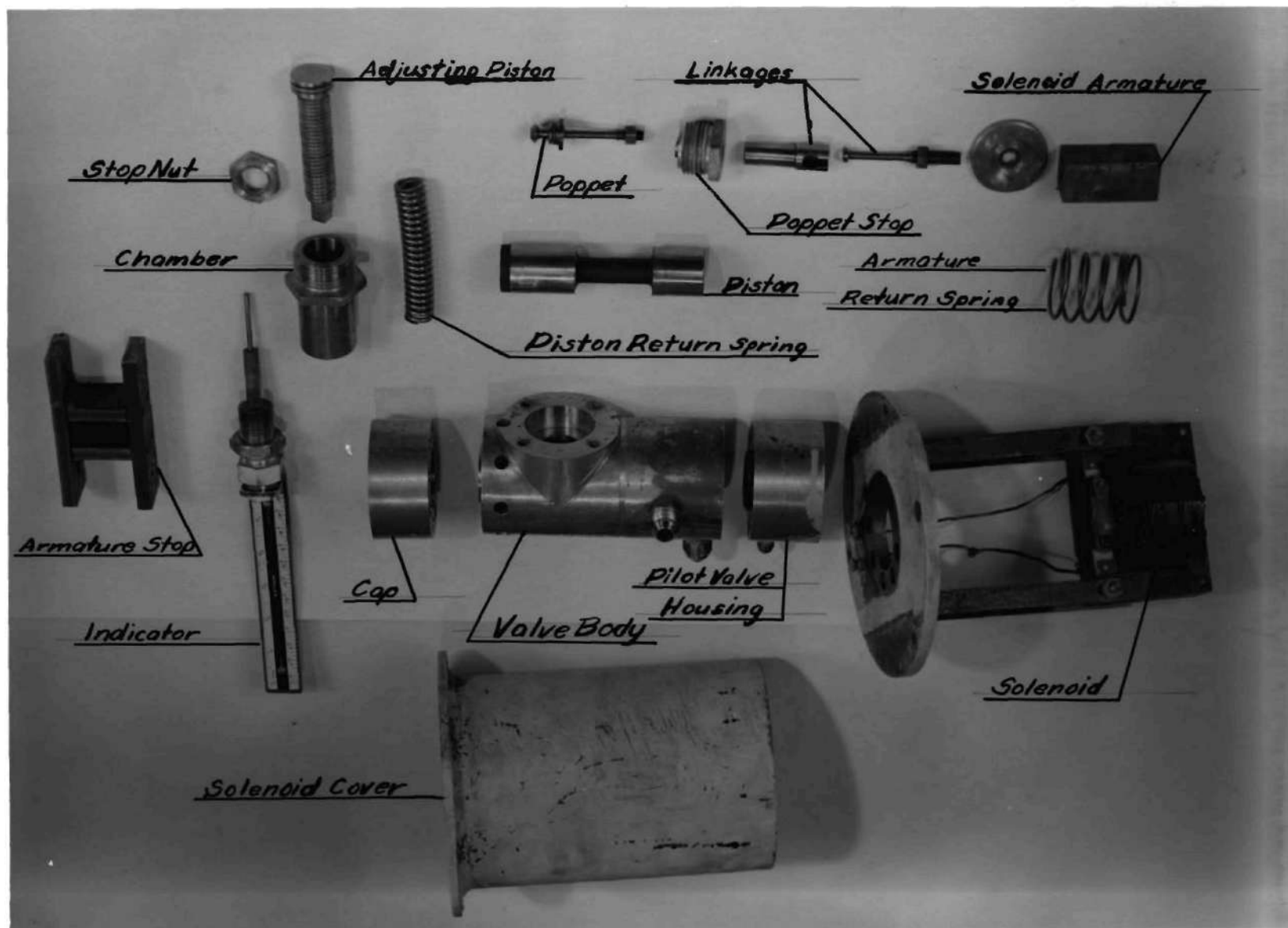


Figure 2. Photograph of Parts

## CHAPTER III

### DESIGN CONSIDERATIONS

Various types of valves might be used to open an area for fluid flow. Several of these, including poppet valves, slide valves, piston valves, and rotary valves, were considered. It was decided that a piston-type valve offered the best possibility of meeting the valving requirements. This decision was based on the ease of machining of the cylinder and piston and the fast opening rates made possible by accelerating the piston prior to uncovering the valve port. In a balanced-piston-type valve the position of the piston has no effect on the internal volume of the valve.

The gas charge was to be confined in the variable-volume chamber and the volume around the minor diameter of the piston. The inlet port was made  $13/32$  inch in diameter to match the inside diameter of the supply line. The bore of the variable-volume chamber was arbitrarily chosen to be one inch. It was desired to have no reduction of area between the chamber and the outlet port. A piston which has a major diameter of  $1-3/16$  inches and a minor diameter section of  $5/8$  inch has an annular area of 0.801 square inches, which is slightly larger than the cross-sectional area of the variable-volume chamber. The discharge port is considered to be fully open when the area uncovered by the

piston is equal to the annular area around the piston. The piston travel past the point of opening to the fully open position is 0.216 inch.

$$\text{Area uncovered} = 0.801 \text{ in}^2$$

$$x(\pi D) = 0.801$$

$$x(\pi \cdot 1.187) = 0.801$$

$$x = 0.216 \text{ inch}$$

$x$  = distance between point of opening and fully open position of the piston

$D$  = cylinder diameter

As illustrated in Fig. 3, the arrangement of various ports requires a minimum amount of piston motion in order to close the inlet port, isolate the charge, and open the discharge port in the proper sequence. The minimum travel of the piston to accomplish this is the minimum travel possible for the piston. This travel is 0.625 inch, of which 0.409 inch occurs prior to the time the piston reaches the point of opening. This part of the piston travel (0.409 inch) is the minimum pretravel which the piston may have. The dynamics of the piston motion as considered later in this thesis indicate that the minimum pretravel is satisfactory for accelerating the piston to the velocities necessary to obtain the desired opening rates of the valve.



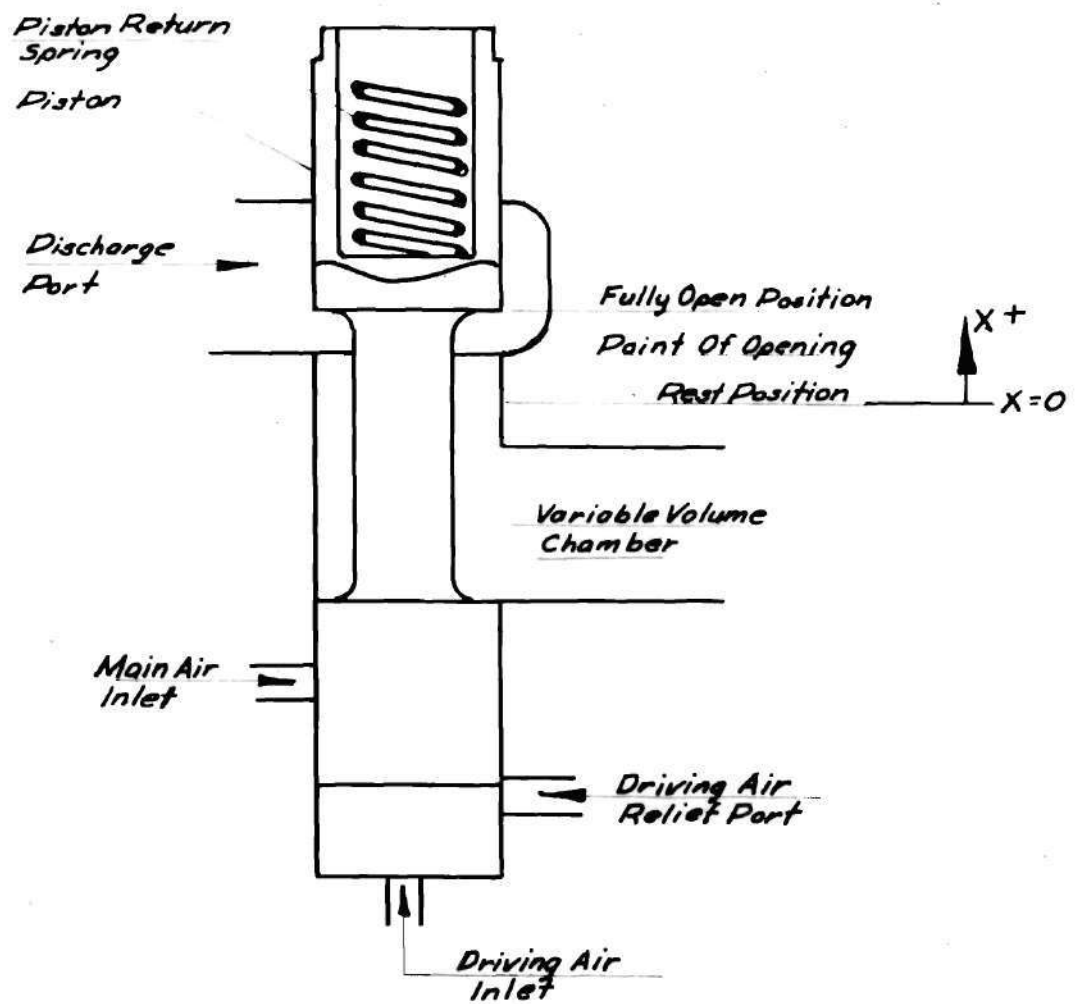


Figure 3. Main Air System

Friction between parts was neglected in the design of this valve. None of the loads applied to the parts is normal to the sliding surfaces. Therefore, the friction forces are relatively small and are thus assumed to be negligible.

Driving air pressure is applied to the head end of the piston to drive the piston through its stroke. As the piston reaches the point of opening, the driving-air relief port is uncovered. The piston is thus accelerated during the 0.409-inch pretravel distance.

After the accelerating portion of the piston stroke, the driving air pressure is relieved. The motion of the piston from the point of opening to the fully open position is a function of the behavior of the gas being released, the rate of decrease of driving air pressure, and the piston-return-spring force. The gas being released will exert a net force tending to decelerate the piston during the interval of time required for a pressure wave to travel from one end of the annular volume around the piston to the other end. The time required for the wave to travel through this distance is small, and the effect of this momentary unbalance may be neglected. The driving air pressure falls off rapidly upon opening of the relief port but continues to drive the piston through a portion of the opening of the discharge port. The piston-return spring exerts a force on the piston which opposes the driving-air-pressure force.

This, then, maintains a force system that is very nearly in balance, so the piston velocity during this portion of its motion was assumed to be constant.

The opening time of the valve was to be controllable over the range from 1/4 to 1 millisecond. The constant velocity required at the point of opening for the piston to uncover the discharge port within the specified limits is as follows:

<u>Opening Time (Milliseconds)</u>	<u>Velocity at Point of Opening (ft/sec)</u>
1/4	72
1	18
from 1/@ point of opening = $\frac{0.216 \text{ in}}{t \text{ sec}} \times \frac{1 \text{ ft}}{12 \text{ in}}$	

In the design calculations the motion of the piston was assumed to be uniformly accelerated during the pre-travel. The accelerations of the piston for the two extremes in opening rate thus are:

<u>Opening Time (Milliseconds)</u>	<u>Acceleration (ft/sec<sup>2</sup>)</u>
1/4	76050
1	4750
from $V_p^2 = 2 A_p \times 0.409 \text{ in.} \times \frac{1 \text{ ft}}{12 \text{ in}}$ , (0.409 in. -- minimum pretravel)	
$A_p = \frac{V_p^2}{2 \times (0.409 \times \frac{1}{12}) \text{ ft}}$	

The pressures required to produce these accelerations were initially estimated using Newton's law,  $\Sigma F = MA$

$$M_p \bar{A}_p = P_{da} A_p - F_{sr}(\text{avg})$$

$$M_p = \frac{1.14}{32.2} \frac{\text{ft sec}^2}{\text{ft}}$$

$$A_p = 1.108 \text{ in}^2$$

$$F_{sr}(\text{avg}) = 32 \text{ lb}_f$$

$$\frac{1.14}{32.2} \cdot \bar{A}_p \frac{\text{lb}_f \text{ sec}^2}{\text{ft}} \frac{\text{ft}}{\text{sec}^2} = P_{da} \cdot 1.108 \frac{\text{lb}_f}{\text{in}^2} \cdot \text{in}^2 - 32 \text{ lb}_f$$

When this expression is solved for the driving air pressure required at the extreme conditions of opening rates, the results are:

<u>Opening Time (Milliseconds)</u>	<u>Driving Air Pressure <math>\frac{\text{lb}_f}{\text{in}^2}</math></u>
1/4	2470
1	180

With this range of driving air pressures, the desired opening rates of the discharge port may be realized with the piston accelerating through the minimum pretravel of 0.409 inch. The design considerations of the motion of the piston indicate that the geometric considerations as described on page six are satisfactory.

After completing the initial design, a more detailed consideration (See the Appendix) was given to the dynamics of the piston. The results of this consideration are

plotted in Fig. 4. The expression for the piston motion during the accelerating part of its stroke is

$$x = \frac{F_{sri} - P_{da}A_p}{K_{sr}} \cos \sqrt{\frac{K_{sr}}{M_p}} t + \frac{P_{da}A_p - F_{sri}}{K_{sr}}$$

This more detailed consideration of the piston motion proved that the driving air pressures required to obtain the extremes of opening rate compare favorably with the calculations made in the initial design.

Opening Time (Milliseconds)	Driving Air Pressure lb <sub>f</sub> /in <sup>2</sup>	
	Initial Design	Detail Consideration
1/4	2470	2250
1	180	180

The piston-return spring had to return the piston to the head end of the cylinder. The return spring was initially loaded to 10 lb<sub>f</sub> to assure that the piston remained against the head end of the valve between strokes. As shown below, a spring with an initial load of 10 lb<sub>f</sub> and a spring rate of 70 lb<sub>f</sub>/in will return the piston in a satisfactory interval of time. The wire size and coil diameter of the spring were then established using the spring data computer.\*

The return time of the piston was initially estimated by assuming uniform acceleration during the piston return

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\*Spring Data Computer, Wallace Barnes Company, Division of The Associated Spring Corporation, Bristol, Connecticut.



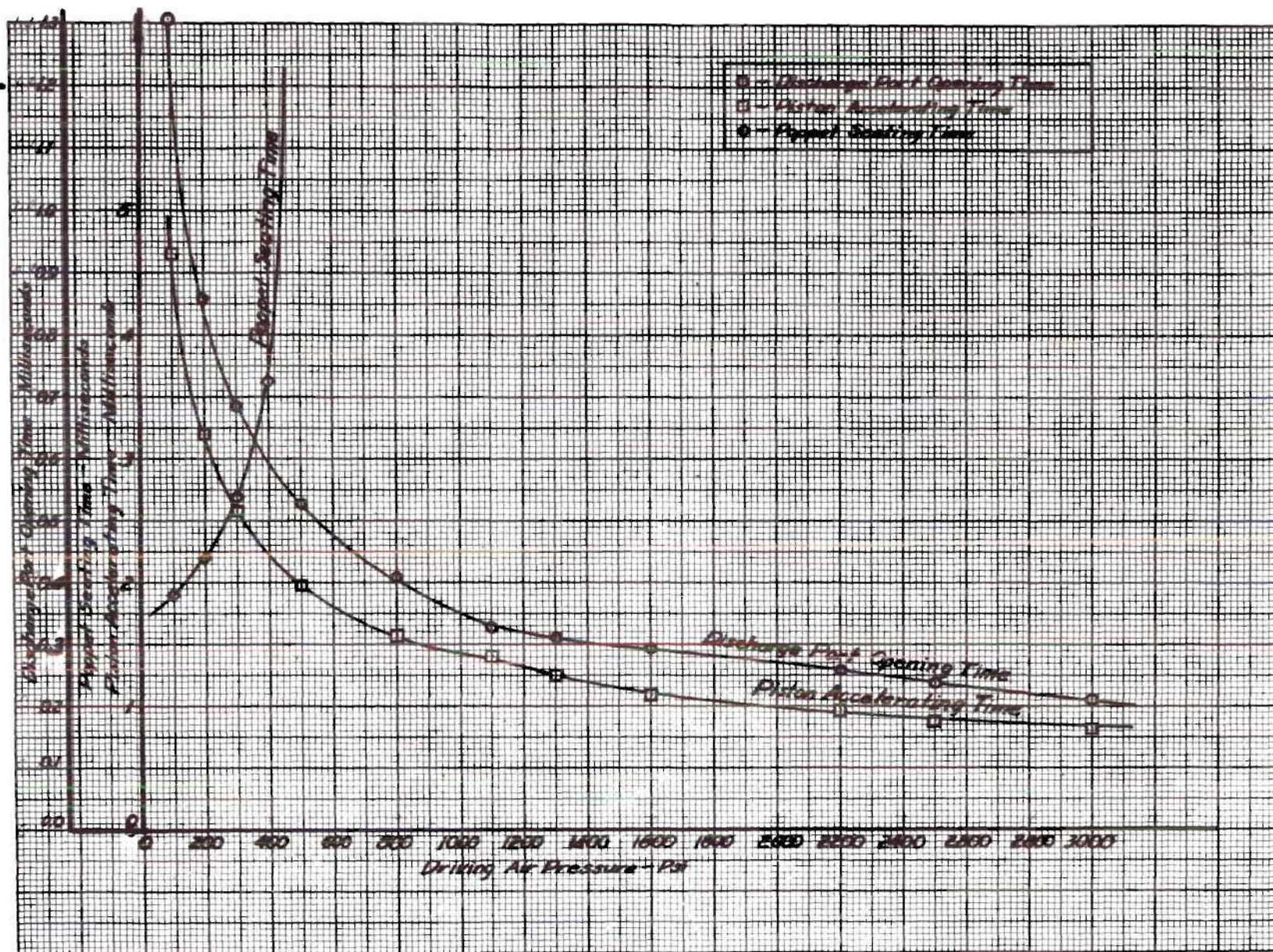


Figure 4. Predicted Valve Performance

350-12 KEUFEL & ESSER CO.  
10 X 10 to the 1/2 inch, 5th lines accented.  
MADE IN U. S. A.



stroke. The return stroke includes the entire distance the piston has traveled away from the head end of the cylinder. This includes the 0.625-inch travel from the rest position to the fully open position and the 0.25-inch penetration of the cushion. The total of 0.875 inch is the length of the return stroke.

$$M_p \bar{A}_p = F_{sr}(\text{avg}) \quad F_{sr}(\text{avg}) = \frac{10 + 0.875 \times 70}{2}$$

$$= 35.6 \text{ lb}_f$$

$$\frac{1.14}{32.2} \cdot A_p \frac{\text{lb}_f/\text{sec}^2}{\text{ft}} = -35.6 \text{ lb}_f$$

$$A_p = -1010 \text{ ft/sec}^2$$

$$x = A_p t^2 \quad x = 0.875 \text{ in}$$

$$0.875 \text{ in} \cdot \frac{1 \text{ ft}}{12 \text{ in}} = 1010 \frac{\text{ft}}{\text{sec}^2} t^2$$

$$t^2 = 0.000073 \text{ sec}^2$$

$$t = 0.0085 \text{ sec}$$

The piston-return time is 8.5 milliseconds which is satisfactory. Subsequent to the initial design, a more detailed consideration of the return stroke of the piston was made and the results compare favorably with the design calculation. The equation of motion of the piston on the return stroke is

$$x = (x_1 - \frac{F_{sri}}{K}) \cos \sqrt{\frac{K_{sr}}{M_p}} t + \frac{F_{sri}}{K_{sr}}$$

When this expression is solved, the return time of the piston is found to be 10.7 milliseconds.

The moving piston has kinetic energy which must be dissipated after the opening of the discharge port without injury to the piston or to other parts of the valve assembly. The use of a water cushion (Fig. 5) presented a convenient method of arresting the piston without the extremely high forces which are coincident with metal-to-metal impact. The valve is to be used while submerged in water. The supply of water for the cushion is the water surrounding the valve; however, if it should become necessary, alterations might be made so that the water could be brought from any convenient source. The design calculations are based on the assumption that the energy absorbed by the cushion may be determined as follows:

$$\text{Energy absorbed} = P_c(\text{avg}) A_p Y.$$

The maximum penetration into the cushion was arbitrarily selected as 0.25 inch. The piston was assumed to decelerate uniformly through the 0.25-inch cushion. The decelerations of the piston and the corresponding times required for the piston to come to rest for the extremes of discharge port opening rate follow:

<u>Opening Time (Millisecond) Discharge Port</u>	<u>Deceleration of Piston ft/sec<sup>2</sup></u>	<u>Stopping Time (Millisecond)</u>
1/4	124000	0.58
1	7780	2.32



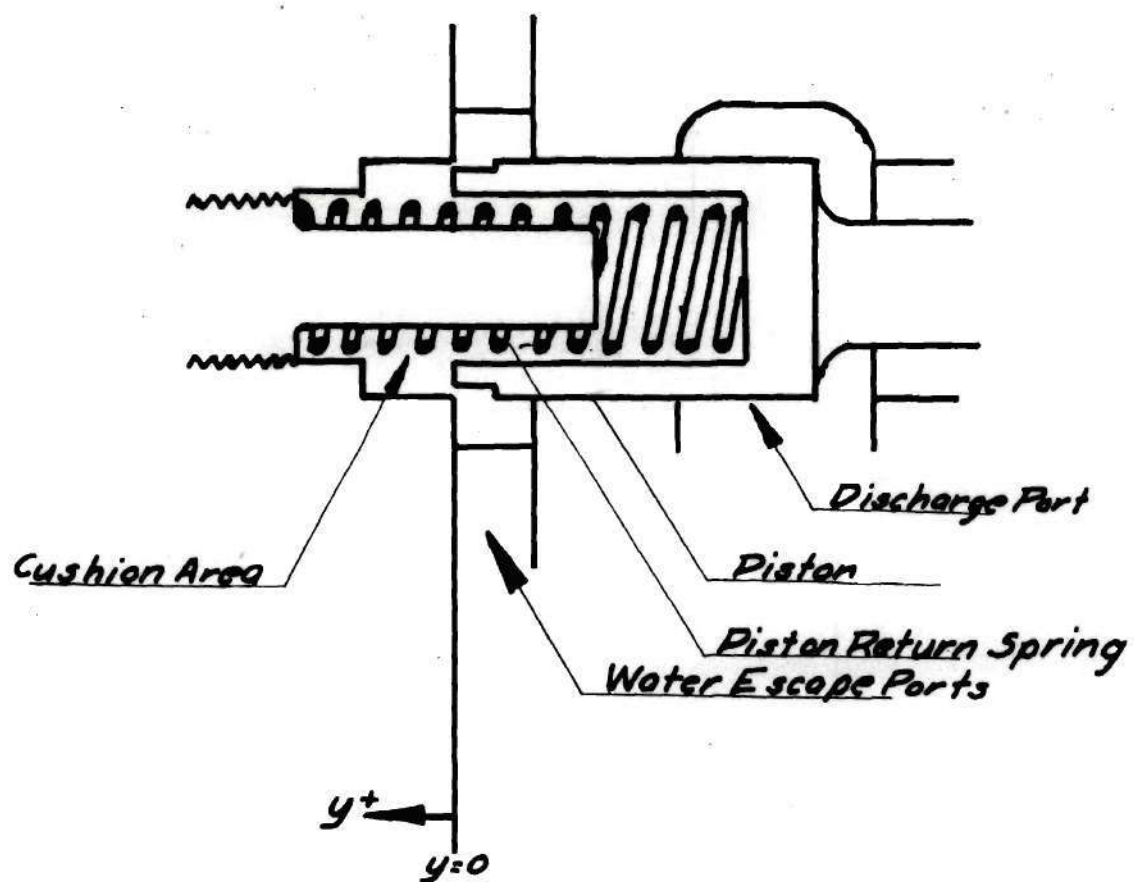


Figure 5. Water Cushion

The compressibility of the water of the cushion is assumed to be negligible compared to the flow of water escaping around the piston. The area of the escape path from the cushion was found by assuming the water escape path to be an orifice with a coefficient of discharge of 0.75. The average pressure in the cushion required to decelerate the piston was estimated as shown below:

$$P_c(\text{avg}) A_p = -M_p \bar{A}_p.$$

For the two opening rates

Opening Time (Milliseconds)	$P_c(\text{avg})$ $\text{lb}_f/\text{in}^2$
1/4	3960
1	249

The area required for escape of the water around the piston is as follows:

For 1/4 millisecond opening time

$$Q = \frac{1.108 \times 2.5 \text{ in}^3}{0.00058 \text{ sec}} \times \frac{1 \text{ ft}^3}{1728 \text{ in}^3} = 0.278 \text{ ft}^3/\text{sec}.$$

$$\text{Also: } Q = A_c V_c = A_c C_d \sqrt{2g b}$$

$$0.278 \text{ ft}^3/\text{sec} = A_c \cdot .75 \sqrt{2 \cdot 32.2 \cdot 3960 \cdot 2.31 \frac{\text{ft}}{\text{sec}^2} \cdot \frac{\text{lb}_f}{\text{in}^2} \cdot \frac{\text{ft}}{\text{lb}_f/\text{in}^2}}$$

$$A_c = 0.000478 \text{ ft}^2 \quad \text{or} \quad 0.069 \text{ in}^2$$

$$b = \frac{A_c}{D}$$

$$b = \frac{.069 \text{ in}^2}{1.187 \text{ in}} = 0.019 \text{ in}$$

<u>Opening Time (Milliseconds)</u>	<u>Ac Regd in<sup>2</sup></u>	<u>Radial Clearance "b" in</u>
1/4	.0690	.019
1	.0695	.019

The radial clearance between the piston and the cushion wall was made 0.020 inch. Subsequent calculations have established that the cushion with 0.02-inch clearance dissipates 97 per cent of the kinetic energy of the moving piston. The clearance was obtained by undercutting the end of the piston which strikes the cushion.

The expression, determined by considerations made subsequent to the design of the valve, which describes the motion of the piston as it moves into the cushion is

$$\ln \frac{V_p}{V_{p0}} = \frac{-C_1 A_p}{M_p} y$$

where

$$C_1 = \left( \frac{A_p}{A_c C_d} \right)^2 \frac{1}{2g} .$$

The pressure in the cushion loads the bolts which hold the cap on the cushion end of the valve body. The maximum pressure in the cushion is assumed to be twice the average pressure in the cushion. For the opening time of 1/4 millisecond, this pressure is 7920 lb/in<sup>2</sup>, and the stress in the 3/8-inch-diameter bolts is 32000 lb<sub>f</sub>/in<sup>2</sup>.

$$\text{Load per bolt} = \frac{7920 \text{ lb}_f/\text{in}^2 \times 1.187 \text{ in}^2}{8 \text{ Bolts}} = 1170 \text{ lb/Bolt}$$

$$\text{Stress} = \frac{P}{A} \cdot f \qquad f = 3.5$$

$$\text{Stress} = \frac{1170 \text{ lb}_f \cdot 3.5}{.110 \text{ in}^2} = 32000 \text{ lb}_f/\text{in}^2$$

The yield strength of the alloy steel bolts used is 125,000 lb/in<sup>2</sup>. The safety factor based on this endurance limit is 3.94. Subsequent calculations show the maximum pressure in the cushion to be 13,550 lb<sub>f</sub>/in<sup>2</sup>. This lowers the safety factor in the bolts to 2.1. Since the initial tightening\* load on each bolt is likely to be, on the average, above the pressure force load, the bolt will not feel the variation in the pressure force.

The pilot valve was to admit driving air to the head end of the cylinder for the time interval required to drive the piston to the point of opening of the discharge port. The geometric arrangement of the pilot valve assembly is shown in Fig. 6. The driving-air relief port in the valve body is located so that the driving air pressure is relieved when the piston reaches the point of opening of the discharge port. In order for the full effect of the driving air pressure to be utilized in driving the piston, the poppet valve should remain open for the time required for acceleration

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\*J. M. Lessells, A. M. Wahl, G. S. Cherniak, Marks' Mechanical Engineers' Handbook, Fifth Edition, Ch. 8. McGraw Hill Book Company, 1951, p. 883

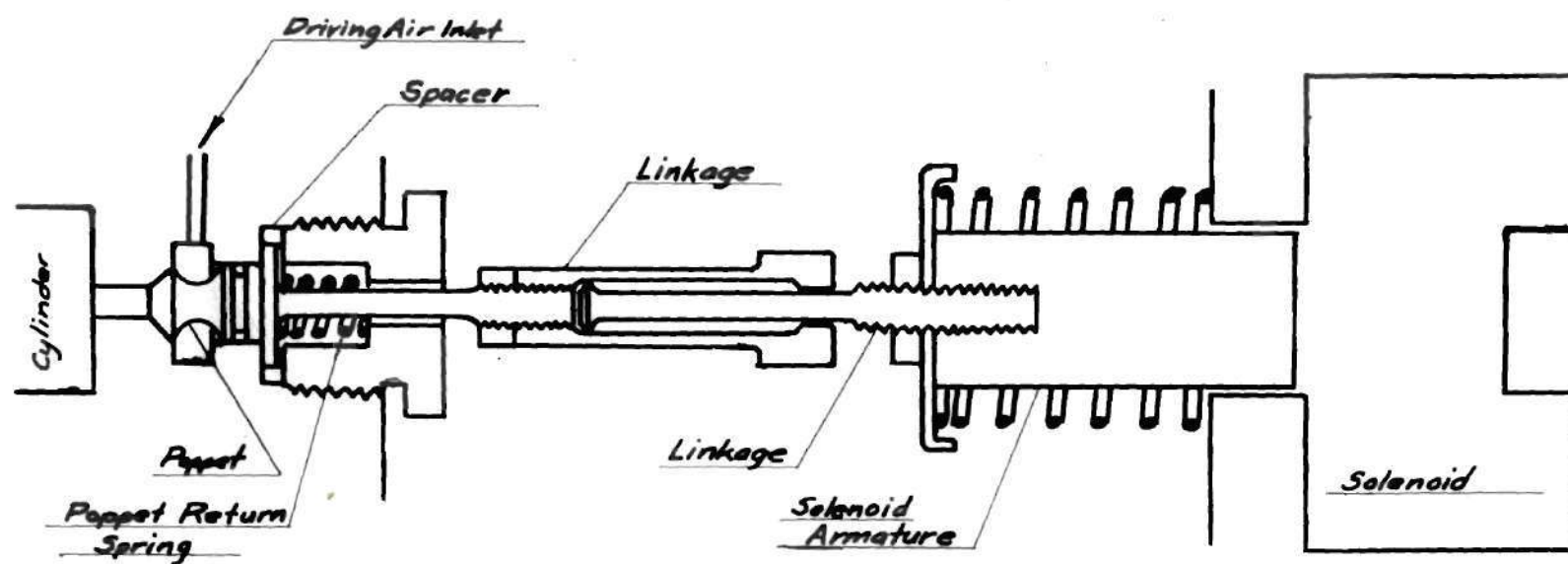


Figure 6. Pilot Valve Assembly

of the piston to the point of opening of the discharge port. The open time of the pilot valve should be independent of the action of the solenoid armature after the initial impact of the linking members.

The poppet valve is opened when the kinetic energy of the moving solenoid armature is absorbed by the poppet-return spring, the poppet stem and the connected links. The solenoid armature bounces clear of the poppet and its attached link when the strain energy of the poppet stem and link is returned to the armature. The poppet-return spring has to accelerate only the poppet and its attached link in reseating the poppet. This action makes the open time of the pilot valve dependent only upon the initial impact of the parts. Of the mechanisms which might have been used, the one used was selected because of its action.

The solenoid used supplies 10 inch-pounds of energy to the armature. The poppet-return spring has an average spring force of 100 pounds and is compressed through  $1/32$  inch. The energy absorbed by the poppet-return spring is 3.12 inch-pounds. The remaining 6.82 inch-pounds are absorbed as strain energy by the poppet stem and the linking members. The endurance limit of the precipitation-hardened stainless steel used for these linkages is 95,000 psi.\* For the purpose of this design, it is assumed that the entire 6.82 inch-pounds appear as strain energy in the poppet stem

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\*Technical Publication No. 2006, American Institute of Mining and Metallurgical Engineers, February 1946.



and the linkage attached to the solenoid armature. The diameter of these two sections is 0.187 inch and their total length is 2.7 inches. The stress in the poppet stem varies from zero to a maximum. For this type of fatigue loading, the allowable stress is 1.5 times as great as the endurance limit.\* The load situation for the members is a fatigue situation; therefore, the maximum stress which may be applied to the members is 1.5 times the endurance limit. The strain energy which may be absorbed by the members is thus considered to be

$$\begin{aligned}\text{Strain energy} &= \frac{1}{2} \frac{A L S^2 L}{E} \\ &= \frac{.0276 (1.5 \times 9.5 \times 10^4)^2 2.7}{2 \times 3 \times 10^7} \\ &= 25.4 \text{ inch-pounds}\end{aligned}$$

The ratio of the energy which may be absorbed to the strain energy expected is

$$\text{Ratio} = \frac{25.4}{6.82} = 3.74.$$

The closing time of the poppet is a function of the mass of the poppet and attached linkage, the poppet-return spring force, and the unbalanced pressure force existing when the poppet is open. The expression for the closing of the poppet follows:

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\*Roark, Formulas For Stress and Strain, First Edition. McGraw-Hill Book Company, 1938, p. 36.

$$F_{pr} - P_{da}A_r = MvAv$$

$$F_{pr} = 100 \text{ lb}_f$$

$$A_r = 0.197 \text{ in}^2$$

$$Mv = 0.00466 \frac{\text{lb}_f/\text{sec}^2}{\text{ft}}$$

Substituting values:

$$100 \text{ lb}_f - P_{da} \times .197 \text{ in}^2 = .00466 \frac{\text{lb}_f/\text{sec}^2}{\text{ft}} A_v.$$

A plot of this expression appears in Fig. 4.

The pressure forces on the poppet are balanced when the poppet is seated. When the poppet is opened, the unbalanced pressure force tends to hold the poppet open. A constriction was placed in the driving-air supply line to cause the pressure in the driving-air system to drop off to a low value when the piston uncovers the driving-air relief port. The poppet-return spring is then able to reseal the poppet.

The open time of the poppet is greater than the accelerating time of the piston for all driving air pressures greater than 290 psi. (Fig. 4). Below this pressure the poppet reseals before the accelerating portion of the piston stroke is complete, and the driving air pressure must be adjusted upward to compensate for the shorter application time of the full driving air pressure.





Figure 7. Tank Installation

## CHAPTER IV

### INSTRUMENTATION AND TEST SET-UP

For the purposes of preliminary testing, the valve was installed in a test tank (Fig. 7) equipped with glass windows in the sides. The downstream piping was left off the valve during the preliminary tests. An indicator was installed in the cushion end of the valve to follow the motion of the piston. A Western Electric Fastax high-speed motion-picture camera was used to record the motion of the piston position indicator and the impact of the pilot-valve parts. The main air and driving air pressures were indicated with Bourdon tube gages.

The operating tests were made with the Fastax camera setup outside the water-filled tank and focused on the piston position indicator. The camera was started and allowed to come up to speed, and then the valve was actuated. A film speed of 2000 frames per second was used. The camera speed was limited by the lighting available for illuminating underwater objects.

The action of the pilot-valve assembly was recorded with the pilot-valve assembly mounted in a vise with the mechanism uncovered. A scale was mounted on the assembly so that the position of the solenoid armature and the linkage connected to the poppet might be measured. High-speed

motion pictures of the mechanism motion were taken at a film speed of 4050 frames per second. No driving air was used for this test.

## CHAPTER V

## TEST RESULTS

It was not possible to plot an actual piston position versus time curve or to describe the type of impact which takes place in the pilot valve. All the parts which were of interest moved so rapidly that a particular position could not be located within the exposure time of the film. The motion picture film gives an estimate of the orders of magnitude of the time in motion of the parts under consideration. These time intervals are in agreement with the time intervals predicted by the preliminary calculations.

A curve of opening rate versus driving air pressure (Fig. 4) was plotted from the detailed consideration made subsequent to the design of the valve. This curve will be used to predict the opening rates of the valve until a more satisfactory method of instrumentation is devised.

## CHAPTER VI

## CONCLUSIONS

The valve as designed meets the requirements originally established. For any given setting of the driving air pressure, the operation of the valve is reproducible.

The valve was designed on the basis of design calculations. Later a more detailed consideration of the motion of the piston and the cushion was made. The detailed considerations confirm the original design.

The comparison between the design calculations and the subsequent detail considerations is shown below.

	<u>Design Calculations</u>	<u>Detail Consideration</u>
Driving Air Pressure for 1 Millisecond Opening Time	180 Psi	180 Psi
Driving Air Pressure for 1/4 Millisecond Opening Time	2470 Psi	2250 Psi
Return Time of the Piston	8.5 Milliseconds	10.7 Milliseconds
Radial Clearance in Cushion	0.02 in	Show that 97 per cent of the kinetic energy of piston is dissipated with 0.02 in. radial clearance

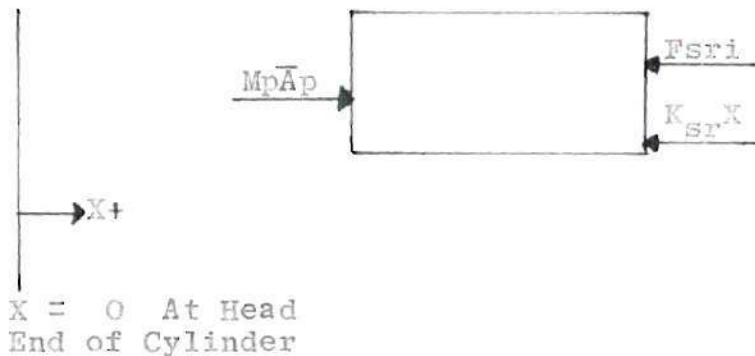
## APPENDIX

## APPENDIX

The motion of the piston is considered for the two periods during which the motion is most critical: (1) The return stroke, which includes the entire return of the piston from the cushion to the rest position with the piston against the head end of the valve, and (2) The accelerating portion of the piston stroke, which includes the motion of the piston between the rest position and the point of opening of the discharge port.

The consideration of the return stroke follows:

The free body of the piston:



The equation of motion:

$$F_{sr1} + K_{sr}X = M_p \frac{d^2x}{dt^2} . \quad (1)$$

The solution of the above equation:

$$X = C_1 \cos \sqrt{\frac{Ksr}{Mp}} t + C_2 \sin \sqrt{\frac{Ksr}{Mp}} t + \frac{Fsri}{Ksr} . \quad (2)$$

When  $t = 0, \quad X = X_0,$

$$X_0 = C_1 + \frac{Fsri}{Ksr} ,$$

$$C_1 = X_0 - \frac{Fsri}{Ksr} . \quad (3)$$

When  $t = 0, \quad \frac{dx}{dt} = 0,$

$$X^1 = -C_1 \sqrt{\frac{Ksr}{Mp}} \sin \sqrt{\frac{Ksr}{Mp}} t + C_2 \sqrt{\frac{Ksr}{Mp}} \cos \sqrt{\frac{Ksr}{Mp}} t, \quad (4)$$

$$0 = 0 + \frac{\sqrt{Ksr}}{Mp} ,$$

$$C_2 = 0.$$

Therefore,

$$X = \left( X_0 - \frac{Fsri}{Ksr} \right) \cos \sqrt{\frac{Ksr}{Mp}} t + \frac{Fsri}{Ksr} . \quad (5)$$

$X_0$  describes the position of the piston at the end of the opening stroke of the valve.

Substituting values and solving for closing time:

$$X_0 = 0.875 \text{ in.}$$

$$Fsri = 10 \text{ lb}_f$$

$$Ksr = 70 \text{ lb}_f/\text{in.}$$

$$Mp = 0.00295 \frac{\text{lb}_f/\text{sec}^2}{\text{in.}},$$

$$X = 0,$$

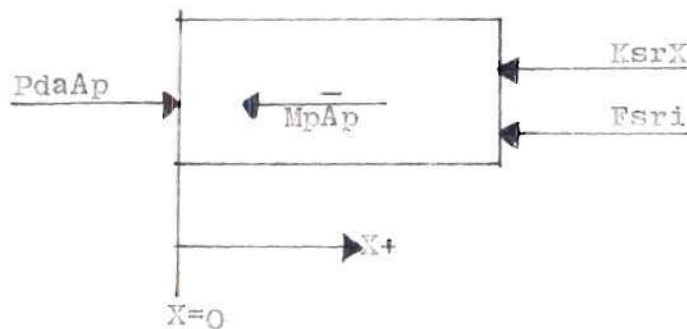


$$0 = \left( .875 \text{ in} - \frac{10 \text{ lbf}}{70 \text{ lbf/in}} \right) \cos \sqrt{\frac{70 \text{ lbf/in}}{.00295 \frac{\text{lbf} \cdot \text{sec}^2}{\text{in}}}} t + \frac{10 \text{ lbf}}{70 \text{ lbf/in}},$$

$$t = 0.0107 \text{ sec.}$$

The consideration of the accelerating portion of the piston stroke follows.

The free body of the piston:



The equation of motion:

$$PdaAp - Fsri - KsrX = Mp \frac{d^2x}{dt^2}. \quad (6)$$

The solution of the above equation:

$$X = C_1 \cos \sqrt{\frac{Ksr}{Mp}} t + C_2 \sin \sqrt{\frac{Ksr}{Mp}} t + \frac{PdaAp - Fsri}{Ksr}. \quad (7)$$

When  $t = 0, X = 0,$

$$0 = C_1 + 0 + \frac{PdaAp - Fsri}{Ksr},$$

$$C_1 = \frac{Fsri - PdaAp}{Ksr}. \quad (8)$$

When  $t = 0$ ,  $x^1 = 0$ ,

$$x^1 = C_1 \sqrt{\frac{Ksr}{Mp}} \sin \sqrt{\frac{Ksr}{Mp}} t + C_2 \sqrt{\frac{Ksr}{Mp}} \cos \sqrt{\frac{Ksr}{Mp}} t, \quad (9)$$

$$0 = 0 + C_2 \sqrt{\frac{Ksr}{Mp}},$$

$$C_2 = 0.$$

Therefore,

$$x = \left( \frac{Fsri - PdaAp}{Ksr} \right) \cos \sqrt{\frac{Ksr}{Mp}} t + \frac{PdaAp - Fsri}{Ksr}. \quad (10)$$

Fig. 4 shows a plot of opening time and accelerating time versus driving air pressure, based on equation 10.

The consideration of the cushion is given below:

$$dVp = \frac{dVp}{dy} dy, \quad (11)$$

$$\frac{dVp}{dy} = \frac{dVp}{dt} \cdot \frac{dt}{dy} = \frac{1}{Vp} \frac{dVp}{dt}. \quad (12)$$

But,  $F = Ma$

and  $PcAp = Mp \frac{dVp}{dt}, \quad (13)$

$$\frac{dVp}{dt} = -\frac{PcAp}{Mp}. \quad (14)$$

Substituting equation 12 into equation 14,

$$Vp \frac{dVp}{dy} = -\frac{PcAp}{Mp} dy. \quad (15)$$

Discharge of water around piston:

$$V_c = \sqrt{2gkD_c}, \quad (16)$$

$$V_c^2 = 2gkP_c. \quad (17)$$

Continuity equation:

$$V_c A_c C_d = V_p A_p \quad (18)$$

$$V_c = \frac{V_p A_p}{A_c C_d} \quad (19)$$

Substituting equation 17 into equation 19,

$$V_p^2 \left( \frac{A_p}{A_c C_d} \right)^2 = 2gkP_c. \quad (20)$$

Let

$$\left( \frac{A_p}{A_c C_d} \right)^2 \frac{1}{2gk} = C_1,$$

$$P_c = V_p^2 C_1. \quad (21)$$

Substituting equation 21 into equation 15,

$$V_p dV_p = - \frac{V_p^2 C_1 A_p}{M_p} dy. \quad (22)$$

The equation of motion of the piston into the cushion:

$$\frac{dV_p}{V_p} = - \frac{C_1 A_p}{M_p} dy. \quad (23)$$

Integrating,  $\ln V_p = - \frac{C_1 A_p}{M_p} y + C_2. \quad (24)$

When  $y = 0$ ,  $V_p = V_{p0}$ , and  $C_2 = \ln V_{p0}$ ,

$$\ln \frac{V_p}{V_{p0}} = - \frac{C_1 A_p}{M_p} y, \quad (25)$$

$$V_p = V_{p0} e^{- \frac{C_1 A_p y}{M_p}}. \quad (26)$$

The cushion as designed had a radial clearance of 0.020 inch. As a check on the rough calculations, the percentage of kinetic energy dissipated by the cushion, based on the above expression (eg.26), follows:

$$C_1 = 0.0186 \frac{\text{lb}_f \text{ sec}^2}{\text{in}^4},$$

$$A_p = 1.108 \text{ in}^2,$$

$$y = 0.25 \text{ in},$$

$$M_p = 0.00295 \frac{\text{lb}_f \text{ sec}^2}{\text{in}},$$

$$V_p = V_o e^{-\frac{.0186 \frac{\text{lb}_f \text{ sec}^2}{\text{in}^4} \times 1.108 \text{ in}^2 \times .25 \text{ in}}{.0295 \frac{\text{lb}_f \text{ sec}^2}{\text{in}}}},$$

$$V_p = V_o e^{-1.75}.$$

For the fastest opening rate,  $V_o = 72 \text{ ft/sec}$ ;

$$V_p \Big|_{y=.25} = 72 \text{ ft/sec} \times .174 = 12.5 \text{ ft/sec}.$$

The percentage of the kinetic energy dissipated:

$$\begin{aligned} \% \text{ dissipated} &= \left[ 1 - \left( \frac{12.5}{72} \right)^2 \right] 100 \\ &= 96.97\% \end{aligned}$$

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